Public Report for ESA-169-3

Company	United States Steel Corporation	**ESA Dates**	September 16 - 18
Plant	Gary Works	**ESA Type**	Pumping Systems
Product	Steel	**ESA Specialist**	Dave Autrey

Summary Report

Introduction: Gary Works is the largest of five integrated steel manufacturing facilities operated by U. S. Steel within the United States. Sheet products, hot strip mill plate products, and tin products are manufactured at Gary Works. The annual raw steelmaking capabilities of the facility exceed 7.5 million tons.

Objective of ESA: The objective of ESA-169-3 was to work with designated U. S. Steel personnel to identify energy saving opportunities in the service water pumping systems that provide cooling water to a large portion of the mill, and to introduce these personnel to the DOE PSAT method of system analysis and opportunity qualification.

Focus of Assessment: Following an exchange of prescreening information via e-mail and telephone, the site assessment team identified service water pumping stations no. 1 and no. 2 as the primary candidate system for this assessment.

Approach for ESA: During the morning of the first day of the assessment, the assessment team discussed the overall layout of the service water system at the facility, and then began a walk-through of the users that are supplied with service water by pumping stations no. 1 and no. 2. During the afternoon the team attended the PSAT webinar on the DOE website as a formal introduction to the PSAT software. Due to the size of the service water system, much of the second day was spent gaining an understanding of how the service water is distributed and utilized. Portable test instruments were then used to take “snapshot” measurements of each operating pump included in the assessment focus. Additionally, logged system operating parameters were obtained from the USS facility control system. On the final day of the assessment, information obtained during the visit was used to demonstrate the PSAT software, and a concise debriefing meeting was conducted.

For calculation purposes the cost of electrical energy, including all demand and surcharges, and taxes, is assumed to be $.05/kWh.

Measured Data

When possible, measured motor power was used as input for calculations. When measured power was not available voltage and current were measured and software estimates of power were used.

In most cases pressure measurements utilized the installed pressure transducers. A few measurements required the use of portable test equipment.

The system is not instrumented for flow rate, and a portable ultrasonic flow meter was not able to make an exact flow measurement. Assuming minimal pump wear/degradation, the speed-adjusted pump head-capacity curve was used to approximate flow rate for each pump. A comparison of this derived flow rate was made to the flow rate determined from the flow-power curve using measured (or, in some instances, calculated) motor power. The relative correlation between these values (and, in some cases, of other related performance indicators) is noted for each pump. It is these estimated flow rates upon which the suggested, potential savings are based.

Discussion and Summary of Potential Opportunities:

Presentation of Supporting Information: Screen captures of PSAT analysis summaries, pump curves, system curves, and other pertinent items of information have been included in the accompanying power point presentation, “ESA-169-3, Attachment A” that was presented to plant management during the ESA exit meeting. Throughout this report, reference will be made to items in this attachment as, for example, “Attachment A.S7”, where the “A” in A.S7 refers to Attachment A, and the S7 refers to slide no. 7.

Service Water Pumping Stations No. 1 and No. 2 – Description of Function

Service water pumping stations no. 1 and no. 2 distribute cooling water from Lake Michigan to an array of cooling equipment and cooling processes throughout the east end of the mill. After providing the necessary cooling, and after being cleaned and reconditioned as necessary, the water is returned to Lake Michigan and/or the Calumet River. Each station includes a battery of relatively high pressure pumps, and a battery of relatively low pressure pumps. The distributed, service water piping associated with the high pressure pumps in each station is connected to that of the high pressure pumps in the other pumping station to provide some degree of interconnectivity and functional redundancy. The low pressure pumps in pumping station no. 1 are primarily dedicated to the condenser of the steam turbines that drive turbo-blower nos. 9 and 10. The low pressure pumps in pumping station no. 2 are primarily dedicated to the STG condenser and other cooling loads associated with the STG.

At the time of the assessment no summary of design or required cooling water flow rates for the service water loads associated with these pumping stations was available. In order to facilitate this preliminary study, a crude attempt to develop such a tabulation has been made. During the assessment walk-through, discussions were held with many, but not all, of the operators of the various cooling process served by these systems. We were unable to gain access and have meaningful discussions with several of the operators of major systems (Q-B.O.P. and No. 2 Casting, for example). Assumptions regarding the service water requirements of major users have been made and included on the partially complete, "Summary of Service Water Loads" that is included (refer to **Attachment A.S3 – A.S5**). It is acknowledged that the listed loads may not be accurate and that many other loads may exist. <u>If this study is pursued further, a more complete and accurate tabulation of service water loads should be compiled.</u> Regarding the attached load summary, note that page 3, space dedicated for tabulation of additional loads, is currently blank. It is included because it contains pertinent notes.

The benchmark against which potential cost savings are compared in this report is, for the most part, the "Estimated Pump Operating Hours & Associated Energy Cost…" that is presented in **Attachment A.S6 – A.S11**. This estimate of the operating cost for PS #1 HP, PS #2 HP, and PS #2 LP pumps was derived from the manual pump logs that are maintained at the plant. The estimated operating cost for PS #1 LP, referred to in subsequent articles, is calculated, in part, from the data collected during the assessment and the estimated flow requirements of condenser no. 10.

A) Pumping Station No. 1 - High Pressure Pumps

The high pressure pumping capacity in pumping station no. 1 is provided by the following pumps.

- Pumps A, B, C, D, E, each of which is an Allis-Chalmers 36 x 30 WSG, horizontally split, double-suction, single stage pump that is rated at 500 RPM with a 46.375 inch (diameter) impeller. Each pump is driven by a constant-speed, 1750 HP, 6900 volt, induction motor.
- Pumps F and G, each of which is a Byron Jackson 56 KXL, single stage, vertical turbine pump that is rated at 585 RPM with a 39 inch (diameter) impeller. Each pump is driven by a constant-speed, 1750 HP, 6600 volt, synchronous motor.

Based upon the available drawings, the discharge water stream from each of pumps A, B, and C are routed, via individual pipes, to the main header in the South Valve House. One branch line connected to each of these discharge pipes prior to the connection of the respective discharge pipe at the main header (for a total of three branch pipes), conducts water to the "T.B. Station Header", which is also located in the South Valve House. The individual discharge pipes from each of pumps D and E terminate in the single header that is located in the North Valve House. One branch line connected to each of these discharge pipes, prior to termination of each main discharge line at the North Valve House header (for a total of two branch pipes), conveys water to the "T.B. Station Header" that is located in the South Valve House. Also, an interconnecting pipe exists between the header in the North Valve House and the main header in the South Valve House.

During the assessment pumps B, C and F were operating.

1) PS #1 – Pump B

Condition A of **Attachment A.S19** displays the PSAT data summary for pump B. **Attachment A.S20** displays the results of the head calculation made by the PSAT head calc tool for this pump. **Attachment A.S23** displays the results of the PSAT system curve tool for this pump. It is important to note that the overall system of which pump B is a part is a relatively complex system, for which the flow v. head characteristics are displayed in other graphs in this report.

Based on the input of measured data, PSAT calculates that the pump is operating at an efficiency of approximately 88.5%. Given that the adjusted manufacturer's performance curve indicates that at the determined operating conditions of flow and head the pump should operate at approximately 83% efficiency, an error in one of the performance parameters seems likely. However, from the PSAT results some very useful information can be obtained.

a. From the “Optimal” column of Condition A it should be noted that pumping equipment is commercially available that will perform the measured duty at an efficiency that is significantly better than the estimated efficiency of the existing pump. At the calculated value of pump operating efficiency, which is likely to be higher than actual, annual energy savings of 859 MWhr (approx. $42,000 @ $.05/KWhr) with an “optimal pump”. Compared to the existing pump operating at the manufacturer’s suggested peak efficiency of 83%, the savings could be substantially more.

b. The existing pump is operating at an output level (1816 hp) beyond the hp rating (1750 hp) of the motor. No service factor was observed on the motor nameplate. A retest of the pump, including the acquisition of an accurate measure of flow rate, at various operating conditions is suggested to confirm this observation. If the pump is, in fact, operating for a substantial number of hours at a motor hp rating beyond a level that is acceptable to the motor manufacturer, then impeller trim is an option that one might consider if the existing pumps are to remain in service for a long period of time. Impeller trim, however, has implications regarding pumps staging and maintenance of system operating pressures.

2) PS #1 – Pump C

Condition B of **Attachment A.S19** displays the PSAT data summary for pump C. **Attachment A.S25** displays the results of the head calculation made by the PSAT head calc tool for this pump. **Attachment A.S26** displays the results of the calculation made by the PSAT system curve tool for this pump. Keep in mind that the overall system of which pump C is a part is a relatively complex system, for which the flow v. head characteristics are displayed in other graphs in this report.

Based on the input of measured data, PSAT calculates that the pump is operating at an efficiency of approximately 82.7%. **Attachment A.S28** indicates that the estimated pump BHP correlates well with that which is projected for the estimated flow rate by the adjusted manufacturer’s power v. flow curve. Similarly, **Attachment A.S29** depicts good correlation between calculated pump efficiency at operating conditions versus projected pump efficiency per the adjusted manufacturer’s efficiency v. flow curve. Based on this degree of correlation between measured vs. projected performance, the following might be worthy of consideration.

a. The estimated flow rate is likely to be somewhat close to the actual flow rate.

b. Given the similarity of operating condition between pump C and pump B, the error in the measured data for pump B that is causing the suspiciously high efficiency estimates may well be power.

c. If the measured conditions are representative of the normal operating conditions, then replacement of the existing equipment with “optimal” equipment could result in savings in the range of 1792 MWhr/yr (~ $89K/yr @ $.05/KWhr).

d. The existing pump is operating at an output level (1935 hp) beyond the hp rating (1750 hp) of the motor. No service factor was observed on the motor nameplate. If the pump is, in fact, operating for a substantial number of hours at a motor hp rating beyond a level that is acceptable to the motor manufacturer, then impeller trim is an option that one might consider if the existing pumps are to remain in service for a long period of time. Impeller trim, however, has implications regarding pumps staging and maintenance of system operating pressures

3) PS #1 – Pump F

Attachment A.S30 displays the PSAT data summary for pump F. **Attachment A.S31** displays the results of the head calculation made by the PSAT head calc tool for this pump. **Attachment A.S32** displays the results of the PSAT system curve tool for this pump. Keep in mind that the overall system of which pump F is a part is a relatively complex system, for which the flow v. head characteristics are displayed in other graphs in this report.

For this example, Condition A of the attached PSAT summary utilizes volts and amps to determine the power that is used in the in determining the efficiency of the pump at the measured operating conditions. The calculated efficiency of 103% is not possible. For this pump an actual power measurement, albeit a reading from a panel mounted analog meter, was available. Using this power measurement, Condition B of the attached PSAT summary determines the efficiency of the operating pump to be 81.6%. Per the adjusted manufacturer’s pump curve the projected efficiency at this operating point, assuming correct flow estimate, is approximately 82.5%. Potential error in reading the analog meter could account for the difference in expected efficiency, as could minor error in flow estimate or head calc. Other factors such as deterioration of pump components (wear rings, impellers) could account for this difference. Nevertheless, the correlation of measured values to those on the adjusted pump curve is good. Condition B of the PSAT summary indicates that if the measured operating condition is representative of that at which the pump performs the majority of the time, then replacement of the existing pump with an "optimal" pump could result in savings in the range of 1400 MWhr/yr (~ $70K/yr @ $.05/HWhr).

4) Comments Regarding System Consumption of Electrical Power and System Flow Rate

Assuming that pumps B, C and F operate at the measured load points for 8760 hours per year, at $.05/KWhr the estimated annual energy cost is approximately $1,834,500 for these three PS#1 high pressure pumps. A brief review of the operating logs reveals that two pumps are operated at times, and at times four pumps are operated. It is worthy of note that pump A appears to draw somewhat fewer amps than other pumps with which it operates in parallel. The very crude estimate of the cost of electricity for the operation of the required PS#1 HP pumps for the year 2007, based on the operating logs, is $1,620,000 (based on $.05/KWhr).

The sum of the flow rates of the users served primarily by PS#1 – HP, and which were visited during the assessment is, as noted in **Attachment A.S4**, approximately 80,000 gpm. These flow rates were determined from information received from the system operators, and from information received by the manufacturers of some of the condensers. The sum of the flow rates being delivered by pumps B, C, and F at the time at which the assessment was performed is approximately 100,000 gpm. Only further study, and possibly flow rate trending if flow meters are installed, will determine if the 20,000 gpm discrepancy between these two sums accounts for the other service water users supplied by this system. The hypothetical load profile that will be referred to in article F of this report, and which will be used to demonstrate potential savings that might be available with a variable water volume system, utilizes a peak service water load of approximately 80,000 gpm for the PS#1 – HP pumps.

5) Opportunities Beyond the Pump House

That portion of the assessment which was spent walking through the various user systems that are served by the high pressure pumps in PS#1 provided insight into several potential opportunities that could improve the thermal quality of the distributed service water, and/or decrease pumping costs. These potential opportunities are briefly mentioned below.

a. Instrumentation Upgrade

It is suggested that consideration be given to installing accurate instrumentation to measure, log, process and distribute information including, but not limited to, the following.

1. Service water flow rates and differential pressures at each service water user.
2. Flow rate, speed, differential pressure, power consumption at each service water pump.

The degree to which various functions are instrumented, and how that information is to be processed, will be dependent upon how the system is ultimately to be controlled (constant or variable volume, constant or variable speed, for example).

b. Service Water Flow Control at BF-4, BF-6 and BF-8

Based upon preliminary, estimated information regarding required service water flow rates and cooling requirements (cooling water flow rate, EWT and LWT) of blast furnaces 4, 6 and 8, a hypothetical service water profile was developed for each of these furnaces based on maintaining a constant water temperature of 82 F at the suction of the cooling water pumps for each furnace. Given that the cooling water systems for these furnaces utilize stand-pipes, implementation of this control would be relatively simple. One might consider directly injecting cooling water into the suction header of the cooling water pumps. In the pipe that delivers the service would be installed a two-way control valve. This valve would respond to changes in temperature of the cooling water for the furnace by increasing or decreasing the amount of service water provided to maintain the cooling water set point temperature. **Attachment A.S36** depicts the hypothetical service water demand profile for BF-4. **Attachment A.S37** depicts the hypothetical service water demand profile for BF-6 and BF-8.

The immediate benefits of implementing these modifications would be a positive impact on maintaining desired pressure in the distribution system, and having a positive impact on the staging of service water distribution pumps. If the overall service water distribution system is ever converted to variable water volume, these modifications will have already been implemented.

c. Operate Fewer Turbo-blower Condenser Pumps When SW Is Sufficiently Cold

Graham Manufacturing provided a partial log of cooling water flow rates that would be required to produce design capacity in turbo-blower condensers no. 9 and 10. The lower end of this range of flows was established based on minimum velocity criteria. The flows that Graham provided were used to develop the service water profile for condenser no. 10 that is depicted in **Attachment A.S38**. As such flow vs. temperature information was not available from Ingersoll Rand, and given the similarity of design between condensers 9 and 10 and condensers 1, 4, 5, 6, 7, 8 (condensers 2 and 3 are out of service), the design cooling water flow rate per horsepower for condenser 10 was used to approximate the design flow rates for each of the smaller condensers. The same flow rate v. temperature equations used to derive the profile for condenser no. 10 could be used to derive a service water flow profile for each of the smaller condensers. For the purposes of this report, the same profile shape and cumulative time characteristics are assumed to apply to all of the condensers.

Two readily apparent benefits of colder service water for the condenser cooling water circuit that is pumped by PS#1 HP pumps are as follows.

1. In the fall, winter and spring of the year, the temperature of the service water will become sufficiently cold so that it is not necessary to admit flow through non-operating condensers in order to sufficiently cool down the water that exits active condenser. The condenser discharge water that is returned to the main service water loop by the circulation pumps will be, for some period of time during the year, sufficiently cool so that its mixing with the service water in the main service water distribution loop does not cause an unacceptable elevation of the temperature of the service water to any users. If fewer condensers are open to flow, then fewer circulating pumps must be run. Based on nameplate pump data, when operating at design conditions, each of the six circulating water pumps that serve turboblower condensers 1 – 8 consumes 131 KW/hr.
2. As can be seen in **Attachment A.S38**, at full condensing load the amount of water required for each condenser decreases significantly as the entering cooling water temperature decreases (Recall that this, and other service water temperature related flow profiles in this report are based on the service water temperature profile shown in **Attachment A.S17**.). It is suggested that since the turboblowers appear to be operated with “on-line” standby capacity (more than the required blowers operate in order to not lose capacity on failure of one blower), then the condensers that are serving the partially loaded blowers require significantly less than design cooling water flow rate for most of the year. If this is, if fact, the operating case, then it is suggested that implementation of cooling water throttling be considered. This would minimize the amount of water used by the condensers and minimize the number of circulating pumps required (One pump could almost certainly serve more than one condenser in many cases.) to operate.

d. Conversion of Distribution System to Variable Water Volume

Conversion of the service water distribution system served by PS#1 – HP pumps to variable water volume has a number of advantages. As these advantages apply to PS# - HP and PS#2 – HP systems, they will be discussed in a single article of this report, article F.

B) <u>Pumping Station No. 2 - High Pressure Pumps</u>

The high pressure pumping capacity in pumping station no. 2 is provided by the following pumps.

* Pumps 10 and 11, each of which is a Goulds model 36 GHC, two-stage, vertical turbine pump that is rated at 710 RPM with a 26 inch (diameter) first stage impeller, and a 24 inch (diameter) second stage impeller. Each pump is driven by a constant speed, 1500 HP, 6600 volt, induction motor.
* Pumps 12 and 13, each of which is a Worthington model 36LN-39, horizontally split, double-suction, single stage pump that was originally rated at 440 RPM with a 39 inch (diameter) impeller. The performance information provided by the client indicates that each of these two pumps has been rerated for 514 RPM with a 45-1/8” (diameter) impeller. Each pump is driven by a constant-speed, 2000 HP, 6900 volt, synchronous motor.

Based on the available drawings, pumps 10 and 11 provide the bulk of the service water to Q – B.O.P. and to other loads to the west of the Energy and Environmental Department. The distribution piping from pumps 10 and 11 also connects with the distribution piping from pumping station no. 1. Pumps 12 and 13 provide the bulk of the service water to Blast Furnace 13/14, North Casting, and other loads to the west of North Casting. The distribution main from pumps 10 and 11 appears to have a cross-connection with the distribution main from pumps 12 and 13, with isolation valves in valve pits 78 and 78A.

During the assessment pumps 12 and 13 were operating.

1) PS #2 – Pump 12

Condition A of **Attachment A.S38** displays the PSAT data summary for pump 12. **Attachment A.S39** displays the results of the head calculation made by the PSAT head calc tool for this pump. **Attachment A.S40** displays the results of the PSAT system curve tool for this pump. It is important to note that the overall system of which pump 12 is a part is a relatively complex system, for which the flow v. head characteristics are shown in other graphs in this report.

Based on the input of measured data, PSAT calculates that the pump is operating at an efficiency of approximately 38.2%. The flow upon which this efficiency calculation is based is an interpolated flow rate from the flow v. head curve, and the digital readout of power that was displayed on the motor controller cabinet. At the operating condition the manufacturer's curve indicates that the pump efficiency should be 43.31%. The lack of close correlation between flow, head and power casts some doubt on the accuracy of some of the measurements. However, it is clear that the pump is operating at flow capacity that is substantially less than its design flow rate of 38,200 gpm. In light of this, it is suggested that consideration be given to the following.

a. The range of performance that is required of pump 12 should be determined. It seems doubtful that the measured condition is representative of its normal operating point (further discussed in next point). If, however, the measured condition is representative of the performance that is expected from pump 12, PSAT indicates that the use of "optimal" equipment could result in savings approaching 7980 MWhr/yr. ($399K @ $.05/KWkr). Even if the design flow rate for the system were the sum of the measured operating points of pump 12 and pump 13, and both pumps are being operated to provide on-line back-up, much more efficient means of addressing this requirement can be utilized. Once the required range of operation is determined, these equipment options should be explored.

b. The preliminary summary of service water loads, reference **Attachment A.S4**, indicates that the combined service water flow rate that PS #2 – HP might be required to deliver is in excess of 70,125 gpm. It is important that this flow rate be accurately determined, and that the various combinations of pumps that are available to address the flow and head requirement be established. Based on the piping drawings it is assumed that pumps 12 and 13 are interconnected to address service water loads together. From the piping drawings it also appears that pumps 10 and 11 can work in parallel with pumps 12 and 13 to address the loads that are connected to the service water distribution piping that originates in Pump Station No. 2. The sum of the flow rates of the pumps 12 and 13 (41,730 gpm) at the time of measurement is not even close to the estimate in the load summary. Furthermore, pumps 10 and 11 were not operating at the time of the assessment. It would seem that at the time of measurement much of the normal service water load addressed by PS #2 HP was not present. It is understood that on and around September 17, 2008, the day that measurements were taken at pumps 12 and 13, production was curtailed in BF #14 and in North Steel in order to accommodate some process equipment maintenance/upgrade. Could this or other possible events be a cause of service water load reduction for PS #2? It is interesting to note that the sum of the design flow rate for pumps 12, 13 and either 10 or 11 is 104,000 gpm. **Attachment A.S41** plots a system curve that includes these parameters.

2) PS #2 – Pump 13

Condition B of **Attachment A.38** displays the PSAT data summary for pump 13. **Attachment A.S44** displays the results of the head calculation made by the PSAT head calc tool for this pump. **Attachment A.S45** displays the results of the PSAT system curve tool for this pump. Keep in mind that the overall system of which pump 13 is a part is a relatively complex system, for which the flow v. head characteristics are shown in other graphs in this report.

Based on the input of measured data, PSAT calculates that the pump is operating at an efficiency of 63.6%. The flow upon which this efficiency calculation was based is an interpolated flow rate from the flow v. head curve, and the digital readout of power that was displayed on the motor controller cabinet. At the operating condition the manufacturer's curve indicates that the pump efficiency should be approximately 60%. While the correlation between flow, head and power is somewhat better than that for pump 12, it is still no so close as to foster solid confidence in the field measurements. As with pump 12, it is quite believable that pump 13 is operating at a flow capacity that is significantly less than its design flow rate of 38,200 gpm. In light of this, it is suggested that consideration be given to the following.

a. The range of performance that is required of pump 13 should be determined. It seems doubtful that the measured condition is representative of its normal operating point (as was discussed for pump 12). If, however, the measured condition is representative of the performance that is expected from pump 13, PSAT indicates that the use of "optimal" equipment could result in savings approaching 3948 MWhr/yr ($197K @ $.05/KWhr). Even if the design flow rate for the system were the sum of the estimated flow rates for pumps

12 and 13, and both pumps are being operated to provide on-line back-up, much more efficient means of addressing this requirement can be utilized. Once the required range of operation is determined, these equipment options can be explored.

b. Refer back to item b. in the above section pertaining to pump 12. The same questions regarding the cause of the relatively low flow rate apply to pump 13.

3) Opportunities Beyond the Pump House

Once the load requirements for the service water pumps in pumping station 2 have been established, a determination can be made regarding whether or not equipment modifications or upgrades will be cost effective. Regardless of the outcome of that investigation, the following opportunities should be considered, as they will enhance the performance and operation of the overall system no matter what equipment is moving the water.

a. Instrumentation Upgrade

It is suggested that consideration be given to installing accurate instrumentation to measure, log, process and distribute information including, but not limited to, the following.

1. Service water flow rates and differential pressures at each service water user.
2. Flow rate, speed, differential pressure, power consumption at each service water pump.

The degree to which various functions are instrumented, and how that information is to be processed, will be dependent upon how the system is ultimately to be controlled (constant or variable volume, constant or variable speed, for example).

b. Conversion of Distribution System to Variable Water Volume

Conversion of the service water distribution system served by PS#2 – HP pumps to variable water volume has a number of advantages. As these advantages apply to PS# - HP and PS#2 – HP systems, they will be discussed in a single article of this report, article F.

C) <u>Pumping Station No. 1 - HP + Pumping Station No. 2 – HP</u>

At the time of this writing, the degree of interconnection between the high pressure service water distribution piping that originates at PS #1 and that which originates at PS #2 is, at least to this evaluator, unclear. How much of each distribution loop can be satisfactorily pumped from its “sister” pumping station should be determined, if it has not already been determined. Given that the service water users are arranged, for the most part, as relatively constant volume users, the primary advantage of establishing the maximum, feasible degree of interconnection between the two distribution systems is potential reliability. Judging by the number of high pressure pumps that were operating in each pumping station during the assessment, 3 of 7 (43%) in PS #1 and 2 of 4 (50%) in PS #2, the service water requirements for this section of the mill appear to be less than the flow rate for which the systems were originally designed. As a result, a generous amount of pump redundancy seems to be available, if the piping is, or can be, arranged to take advantage of this windfall. It is suggested that field testing for establishing the level of interoperability that exists between these two pumping stations be designed and performed. The results, if satisfactory, might be documented and incorporated into the operating procedures for the systems.

A substantial degree of interconnection between the two service water distribution systems becomes a significant advantage if the system is converted to variable water volume (refer to article F). As the lake water becomes colder and the overall requirement for service water decreases commensurately, line losses diminish and it becomes more feasible for the pumps in either pumping station to serve loads at any of the distant service water users. With proper system instrumentation, it then becomes feasible to automatically stage the most efficient combination of pumps in one or both pumping stations to provide service water with minimum electrical cost.

D) <u>Pumping Station No. 1 - Low Pressure Pumps</u>

The low pressure pumping capacity in pumping station no. 1 is provided by the following pumps.

* Pumps 1B, 2B, and 3B, each of which is a Wilson Snyder model 28 MKL, single-stage, vertical turbine pump that is rated at 1180 RPM. Each pump is driven by a constant-speed, 250 HP, 460 volt, induction motor. As the high pressure pumps were the focus of the assessment, no performance curves were immediately available for these pumps.

Based on the available drawings, pumps 1B, 2B and 3B are dedicated primarily to the pumping of service water through the condenser of the steam turbine for turbo-blower condenser no. 10, with the service water leaving the condenser being routed to GW7. Two of the pumps operating in parallel are required to provide the design flow rate of 19,000 gpm to the condenser. One pump serves as a spare.

During the assessment pump 2B was operating.

A pump curve for these three, identical, vertical turbine pumps was not available at the time of the assessment. For purely illustrative purposes, a Goulds vertical turbine pump was selected for the design duty (9500 gpm @ 85 ft. w.c.) of each of the installed pumps. For this illustration, the system curve was developed using 33 ft. as the head at 0 flow, and 85 ft. as the TDH at 19,000 gpm.

For this example, in the absence of a flow trend, information was obtained from Graham Manufacturing, the manufacturer of the condenser, that states the design cooling water flow rate for condenser no. 10 to be 19,000 gpm at 75 deg. F. Graham also provided the required flow rates for producing 100% condenser capacity down to an entering water temperature of 35 deg. F. **Attachments A.S49** and **A.S50** are the service water demand profile that resulted from this flow information was correlated to the lake water temperature profile. **Attachment A.S51** shows the results of the pump energy calculations associated with this flow profile. In light of this illustrative information, it is suggested that consideration be given to the following.

a. USS service water system operators are obviously aware of the opportunity that colder water and a partially loaded condenser present for pump energy savings. Only one of the three pumps was operating at the time of the assessment.
b. An NPSH curve for the existing pumps should be examined to determine the suitability of the existing equipment for single pump operation.
c. With and accurate load profile, system curve, and pump curves, an investigation should be made to determine the most efficient staging points for the existing pumps. This could be the basis for a pump staging algorithm that operates the most efficient combination of pumps needed to maintain the desired condenser pressure at various entering water temperatures.
d. Based on the illustrative operating cost summaries in **Attachment A.S51**, which assumes that a flow required for 100% condenser capacity is required at all times, the estimated electrical power consumption for staged, constant speed pumps is approximately 1985 MWhr/yr ($99K @ $.05/KWhr.), and 1306 MWhr/yr ($65K @ $.05/KWhr). The savings for using optimally staged, VFD controlled pumps could be as much as 679 MWhr/yr ($34K @ $.05/KWhr).
e. Another potential opportunity that becomes apparent is that when the service water is sufficiently cold, consider allowing the PS #1 low pressure pumps to provide cooling water to condensers no. 9 and 10, in lieu of providing service water to condenser no. 9 from the high pressure pumps. There will exist a point at which the power required to deliver the reduced flow of water to the two condensers will be less than the power required to pump water to condenser no. 10 and operate the 300 hp (est. actual KW use, 209 KW). One process factor that should be considered, especially if the users are not converted to variable water volume, is that below a certain temperature it may be desirable to add the heat from condenser no. 9 to the service water.

E) <u>Pumping Station No. 2 - Low Pressure Pumps</u>

The low pressure pumping capacity in pumping station no. 2 is provided by the following pumps.

* Pumps 17, 18 and 19, each of which is an Ingersoll Rand model 47 PKM, single-stage, vertical turbine pump that is rated at 718 RPM with a 34 inch (diameter) impeller. Each pump is driven by a constant-speed, 1500 HP, 6600 volt, induction motor.

Based on the available drawings, pumps 17, 18 and 19 are dedicated primarily to the providing service water to the STG and to the Electric Power Station #5. Two of the pumps operating in parallel are required to provide the design flow rate to the STG condenser. Other than the flow rate going to the STG condenser, it is not clear, at least to this evaluator, what other loads are served by these pumps. One pump serves as a spare.

During the assessment pumps 17 and 18 were operating.

1) PS #2 – LP – Pump 17

Condition A of **Attachment A.S52** displays the PSAT data summary for pump 17. **Attachment A.S53** displays the results of the head calculation made by the PSAT head calc tool for this pump. **Attachment A.S54** displays the results of the PSAT system curve tool for this pump. It is important to note that the overall system of which pump 17 is a part may be a relatively complex system, for which the flow v. head characteristics are displayed in other graphs in this report.

Based on the input of measured data, PSAT calculates that the pump is operating at an efficiency of approximately 88.6%. The manufacturer's curve indicates that at the measured operating conditions the pump should perform at approximately 91.9%. Given this good correlation between efficiency and power, the flow rate that was interpolated from estimated TDH and the flow v. head curve is believable. Furthermore, little, if any, benefit would be derived from the installation of “optimal” equipment. System opportunities will be discussed in paragraph 3 of this article.

2) PS #2 – LP – Pump 18

Condition B of **Attachment A.S52** displays the PSAT data summary for pump 17. **Attachment A.S58** displays the results of the head calculation made by the PSAT head calc tool for this pump. **Attachment A.S59** displays the results of the PSAT system curve tool for this pump. It is important to note that the overall system of which pump 18 is a part may be a relatively complex system, for which the flow v. head characteristics are displayed in other graphs in this report.

Based on the input of measured data, PSAT calculates that the pump is operating at an efficiency of approximately 91.7%. The manufacturer’s curve indicates that at the measured operating conditions the pump should perform at approximately 92.4%. Given this very good correlation between efficiency and power, the flow rate that was interpolated from estimated TDH and the flow v. head curve is believable. Furthermore, little, if any, benefit would be derived from the installation of “optimal” equipment. System opportunities will be discussed in paragraph 3 of this article.

3) Opportunities for Performance Enhancement

a. For illustrative purposes only, a hypothetical cooling water demand profile was developed for the STG condenser. This profile was developed using the lake water temperature profile, the condenser flow v. cooling water temperature relationship for condenser no. 10, and the power generation trend provided by USS for most of 2007. Regarding this hypothetical profile, which is shown in **Attachment A.S63**, note the following.

   1. The profile is a very rough approximation that is based on a very specific set of data for a particular operating period. As levels of power generation, lake temps, degree of tube fouling change, required cooling water flow rates will change. These factors will, in all probability, result in a very different profile.
   2. The profile does not account for any by-pass circuits.
   3. The profile assumes that all cooling water loads served by pumps 17, 18, 19 have the same cooling water demand profile.
   4. The estimated cost of electrical power that is consumed by the service water pumps is assumes that system curve “B” shown in **Attachments A.S55 and A.S60** remains fixed.

b. Estimated electrical power cost for staging the existing, constant speed pumps to meet the estimated STG cooling water loads based on this profile wad $881,850/year. This cost estimate compares favorably to the $896,595 energy cost estimate that was generated from the log sheets (see **Attachment A.S11**). The estimated electrical power cost for meeting this water demand with pumps controlled via VFDs is approximately $504,457/year. As compared to the model-based, constant speed pump energy cost, the VFDs offer a potential annual savings in the range of 7,547 MWhr/yr ($377K).

c. It is suggested that, among other potential energy saving option for this system, the owner give consideration to the following.

   1. First and foremost, develop an accurate profile of the required flow rate vs. lake water temperature for this system. If this profile supports some type of flow modulation, proceed with investigating these opportunities:
   2. Determine the cost effectiveness of operating a single pump during those hours when one pump can provide adequate flow. Careful consideration should be given to the NPSH requirements of the pump, and also to relatively inexpensive means of automatic modification of system flow v. head characteristics so that a solo pump will remain at a safe operating point in its flow v. head curve.
   3. Determine the cost effectiveness of installing a smaller, “jockey” pump that can be operated in parallel with one of the larger pumps to provide additional flow during those times when two large pumps provide far too much flow, and one large pump does not provide enough flow.

F) <u>Variable Water Volume at Service Water Users</u>

Assuming a constant heat exchanger load at each service water user (make-up water functions excluded), as the temperature of the service water becomes colder, less of it will be required to provide the same heat rejection, assuming that an increase in water temperature rise is acceptable. There is for any heat exchanger a lower limit to

the flow rate that can be tolerated. For the purposes of this report, a conservative, lower limit of cooling water flow rate that might be permissible has been assumed to be 70% of design (or a 30% flow reduction). Furthermore, it has been assumed that for each of the users in this system, maximum design service water flow rate requirement occurs at a service water temperature of 80 F. The minimum flow rate is assumed to occur at 50 F. If the required cooling water flow rate is assumed to decrease from 100% "design" @ 80 F to 70% of "design" at 50 F service water temperature, a hypothetical service water load profile that takes into account the lake temperature profile can be generated. Such a profile was generated for the PS #2 HP, and PS #1 HP systems. A condensed version of such profile is included in **Attachment A.S64.** Note that the illustrated profile is for PS #2, but the shape of the similarly based profile for PS #1 is identical, with the incremental flow rates being commensurately larger.
Note that the comments and estimated savings presented herein assume that the hydronic cooling loops at the service water users remain "open".

1) Benefits of Variable Water Volume
   a. Service Water Pumping System
      1. Reduced pumping costs (Flow rate changes with load, opportunity to operate at lower pressures for many hours of the year)
      2. Improved redundancy at loads other than design loads
      3. Improved pumping equipment life and reliability
   b. Service Water Users

      The advantages stated above for the service water pumping system apply to the pumping systems that serve the service water users.
2) Disadvantages of Retrofitting for Variable Water Volume

   The only disadvantage of retrofitting the system for variable water volume is capital cost. At initial construction, variable water volume is usually very competitive in first cost with constant volume systems. However, it simply cost money to retrofit an existing system.
3) Methods of Implementation

   Several basic methods of instituting variable volume control of the service water at the users include, but are not limited to:
   a. Pump staging
   b. Throttling valves + pump staging
   c. VFDS + pump staging
   d. Zone/distributed pumping + VFDs
4) Estimated Savings
   a. Service water pumping systems

      Modeling of the existing pumps controlled via VFDs to meet the demands of a load profile with the characteristics of that shown in **Attachment A.S64** suggests that with a 40 psig base system pressure savings in the range of 20% of the cost of a similarly staged, constant speed pumping system are available. In the case of PS #1, such savings could be in the range of 6,480 MWhr/yr ($324K/yr.). In the case of PS #2 such savings could be in the range of 3,430 MWhr/yr ($171K/yr.). The savings can become even larger as decreased water flow requirements at reduced supply water temps make it possible to operate the service water distribution system at reduced pressures.
   b. User Pumping Systems

      In many cases, similar savings in pumping energy cost are attainable in each of the user pumping systems.

      It is important for the reader of this report to realize that the above asserted savings are not presented as guarantees. They have been generated in a hypothetical context due to the lack of factual load profiles, but they are based on sound engineering principles and experience to illustrate the relative magnitude of savings that might be achievable with a variable water volume distribution system.

G) <u>Closure of Hydronic Cooling Loops at Users</u>

Making the hydronic cooling loops at each of the users closed in lieu of open is not an inexpensive retrofit, but it is suggested that the following advantages and disadvantages be given careful consideration. Such an upgrade is not an item of "low hanging fruit", but rather and investment that has long-term benefits. It is assumed that each of the cooling loops would be designed for variable water volume.

1) Advantages

The following list of advantages includes some that are true for all closed loop systems, and some that apply specifically to this closed loop system.

a. Lower service water pumping costs, in this particular case, because the differential pressure that the service water pumps would have to provide to drive flow across the interfacing heat exchangers would be less than the pressure that is currently required at each user interface.
b. In this case, a contribution to lower service water pumping cost results from the fact that as the lake water temperature becomes colder, its flow rate can be throttled to that flow rate required to facilitate proper heat transfer without having to maintain minimum flow rates required by directly connected, open system users.
c. Lower electrical cost for the user circulating pumps due to the elimination of static head - In many cases, the static head that is being eliminated is equal to or greater than the service water pressure that is currently required at the suction header of the user pumps. Thus, in many cases, the existing pumps and motors may work in the closed loop. Additionally, the elimination of static head substantially reduces part load energy consumption for VFD controlled pumps serving a closed loop.
d. Reduced chemical treatment costs.
e. With proper chemical treatment, scaling and corrosion of interior heat exchange surfaces are eliminated. Additionally, biological fouling or interior heat exchange surfaces is not a problem.
f. More accurate and sensitive leak detection systems can be employed.

2) Disadvantages

a. Capital cost - Building a closed loop system from the ground up, especially in a case where a source of cooling water exists, is usually comparable in cost to an open loop system. In the case of a retrofit, however, additional capital cost is a larger consideration.
b. Higher temperature of entering cooling water temperature – Due to the fact that a heat exchanger usually exists between the primary cooling medium and the closed loop cooling medium, the temperature of the cooling medium in the closed loop is usually higher than that of an open loop. However, in a properly maintained closed loop, the interior of the heat exchange surfaces will experience less fouling than the interior heat exchange surfaces in an open loop. This can offset some of the lower LMTD.

Management Support and Comments:

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